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ENGINE

This invention relates to engines using external sources of heat to perform useful work.

It is known that useful work can be performed by utilising sources of heat that are external to the working apparatus. For example, steam engines and Stirling engines use external sources of heat to drive reciprocating pistons by utilising the increase in pressure and gas expansion of a working fluid. However, there are available sources of heat energy that have either not been effectively utilised to perform useful work or for which an alternative engine to utilise the energy to perform work would be useful.

It is an object of the present invention to provide an engine which can utilise external heat sources to perform useful work.

The engine according to the present invention has been developed so as [illegible] possible the use of relatively low temperature sources of heat energy or to utilise heat energy from sources that is presently being wasted. However, it will be appreciated that the invention is not limited to energy sources of low temperature. Fluid with temperatures of up to 100°C and beyond could be utilised by the invention. The temperature of the energy source may determine the type of thermal energy converter, such as an evaporator and condensor, which may be used. It may also determine whether the converter components are to be freestanding or integrated in the form of a converter module.

For example, solar radiation can be used to readily heat water to modest temperatures such as 40°C-60°C and it would be advantageous if such heat energy could be used to perform useful work. Heated water or heated water vapour can be obtained from hydrothermal sources. For example, bore water extracted from ground aquifers and used for

irrigation or for drinking water for stock or for domestic use in remote locations is often at an elevated temperature and it would be beneficial if the heat energy of such water could be utilised to perform useful work. Also there are many possible sources of heat energy that are presently unutilised or underutilised such as heat energy in exhaust gases from internal combustion engines such as engines driving generators or even being used in vehicles, heat energy in exhaust gases or smoke being discharged from industrial plant and equipment and it would be beneficial if such heat sources could be utilised to perform useful work.

The engine according to the present invention includes an expansion chamber having a moveable wall so that the chamber has a variable volume, the chamber containing a working fluid which increases in pressure and expands upon being heated so as to move the wall upon being heated and increase the volume of [illegible] nd conversely which contracts upon being cooled so that the wall mov[illegible] e direction to decrease the volume of the chamber. The expansion chamber may be defined by a cylinder having a piston moveable therein, the piston defining the moveable wall of the chamber. The working fluid may be any suitable material such as a refrigerant of the kind used in refrigeration and air conditioning plant, e.g. freon gases.

The engine also includes a fluid heating means for applying heat from an external source to the working fluid during a heating cycle of the engine so as to heat the working fluid causing expansion of the working fluid in the expansion chamber. The engine further includes fluid cooling means for cooling the working fluid during a cooling cycle commencing after the heating cycle so as to cause contraction of the working fluid in the expansion chamber and decrease of the volume thereof. The fluid heating means and fluid cooling means may include a heat exchanger for supplying heat energy to the working fluid

and for extracting heat energy from the working fluid, the heat exchanger being provided with heat energy from the external source during the heating cycle, e.g. by being supplied with heated water from solar heat collectors or by thermal ground water or directly or indirectly with heat from a source of waste heat energy. Conversely during the cooling cycle, the heat exchanger may be supplied with a cooling medium such as surface water from any convenient local source.

The engine further includes means for cycling the fluid heating means and fluid cooling means alternately so as to alternately heat and cool the working fluid and cause reciprocating motion of the moveable wall of the expansion chamber. The means for cycling may be means for switching the supplies of heating medium and cooling medium to the fluid heating me[illegible] cooling means in alternating fashion synchronised with, e.g. in response [illegible]ment of the moveable wall of the expansion chamber reaching predetermined points in its reciprocating movements.

The engine further includes pressure storage means operatively associated with the moveable wall of the expansion chamber and including compression means coupled to the moveable wall for compressing a storage fluid during one of the cycles of the moveable wall and accumulator means for holding pressurised storage fluid at an elevated pressure and at progressively increasing pressure as the compression means cycles in response to cyclical movement of the moveable wall of the expansion chamber. Because the accumulator means stores pressurised storage fluid at an elevated pressure, it is capable of performing useful work and therefore the engine is in use operatively associated with a controlled work output system for utilising the stored pressurised storage fluid to perform useful work.

The compression means of the pressure storage means may comprise a piston movable within a cylinder, preferably a piston of substantially smaller diameter than the diameter of the preferred moveable piston in the expansion chamber so that the working fluid pressure developed in the expansion chamber is magnified because of the step down ratio of piston areas. The compression piston moveable in the compression chamber in response to movement of the moveable wall of the expansion chamber, compresses the storage fluid. The storage fluid being compressed by the compression means can be supplied to the accumulator means so as to progressively increase the pressure and volume of the fluid held by the accumulator means.

The storage fluid may be an oil and the accumulator means may be oil accumulators [illegible]lly known type used for storing hydraulic oil at elevated pressure for subsequent [illegible]d release. The controlled work output system may for example include an hydraulic motor through which the pressurised stored hydraulic oil can be released in a controlled manner so that the hydraulic motor can perform useful work, e.g. by being coupled to an alternator or generator to produce electrical energy for direct utilisation or for charging storage batteries. The storage fluid, which in the preferred embodiment is hydraulic oil, can be returned after being released from the hydraulic oil accumulator means, through the hydraulic motor, to a reservoir. The storage fluid held at low pressure in the reservoir can be progressively drawn into the compression means during a return stroke of the compression piston, whereas during the compression stroke of the compression means the supply line to the reservoir is closed and the hydraulic oil being compressed is supplied to the oil accumulator means.

In another aspect the invention provides an engine for converting thermal to stored energy for doing work, the engine including:

(a) a thermal engine converter including a first converter or expansion chamber having a first movable converter wall together defining a first converter compartment of variable volume, the first converter wall capable of performing a stroke cycle;

(b) a working fluid in said first converter compartment which expands upon being heated and contracts upon being cooled whereby to displace the first converter wall;

(c) a temperature modifier:

(i) to heat the working fluid to expand the volume of the first converter compartment by displacement of the first converter wall as a first part of the stroke cycle; and

(ii) to cool the working fluid to reduce the volume of the first co compartment and to permit the return of the first converter wall as a second part of the stroke cycle;

(d) pressure storage means operatively associated with the first converter wall and adapted to deliver pressurised fluid to an accumulator;

(e) said accumulator for storing the pressurised fluid at an elevated pressure; and

(f) controlled work output means for converting the energy associated with the pressurised storage fluid to a useful form.

Preferably the controlled work output means is capable of energy conversion at a constant rate, irrespective of the rate of the stroke cycle or of the first or second part of the stroke cycle. For example, the stroke cycle of the first converter wall may be irregular such that stroke cycles may vary in the total time required to complete a stroke cycle. Moreover,

the actual stroke of the first converter wall may be of non-uniform speed. Indeed, typically there is resistance to the travel of the first converter wall during the first part of the stroke cycle and that resistance is variable over time. The first converter wall may accelerate during its travel through the stroke as the resistance dissipates. On the other hand, where there is little or no resistance to the travel of the first converter wall during the stroke, such as where the opposed converter compartment is vented to the atmosphere, the first converter wall may experience high initial acceleration, followed by steady deceleration as the first converter compartment expands thereby effectively decreasing the pressure in the first converter compartment. Irrespective of the rate of the stroke cycle, however, the pressure storage means may be effective to ensure delivery of sufficient pressurised fluid to the accumulator to enable the accumulator to supply the co[illegible] output means at a [illegible] constant rate, if required.

The first expansion chamber and the first converter wall may together define a first opposed converter compartment within the first expansion chamber. The first opposed converter compartment may be of variable volume. Preferably the first expansion chamber as a whole defines a chamber with a constant volume such that the variable volume of the first converter compartment is in inverse relationship to the volume of the first opposed converter compartment.

The working fluid preferably has a high thermal expansion co-efficient. The working fluid may be a gas or liquid. Preferably the working fluid is a liquid. Even more preferably, the liquid is a refrigerant such as freon. In a particularly preferred embodiment, the working fluid is refrigerant type AZ20.

The temperature modifier includes any suitable means for heating the working fluid. For example, the temperature modifier may include an evaporator supplied by an independent source of thermal energy. The independent source of thermal energy may be solar heat collectors, thermal ground water, hot waste gas or liquid from a power plant etc. used to supply an evaporator by known means. The condenser may draw on locally available cool fluid sources such as water streams, dams and other cool fluid sources.

The pressure storage or intensification means may include a hydraulic or pneumatic arrangement. Preferably, the pressure intensification means includes a hydraulic arrangement. The pressure intensification means may include the first opposed compartment and the pressurised fluid may, in part, occupy the space in the first opposed compartment. The first cc [illegible] y include a first face facing the first expansion compartment and may in [illegible] face facing the first opposed compartment. The first face may be significantly greater in surface area than the second face to achieve the pressure intensification desired between the thermal energy converter and the pressure intensification means. The available volume to the pressurised fluid in the first opposed compartment may be reduced by the presence of a column occupying space between the second face and an aperture in the end wall of the first opposed compartment through which the column may extend. The column may be any suitable configuration or orientation within the first opposed compartment, provided that it has a constant cross-section throughout its length or the section of length adapted to travel through the aperture.

In another arrangement, the pressure intensification means may include a first pressure chamber housing a first pressure chamber wall. The first pressure chamber wall may be movable to define a first pressure chamber compartment of variable volume. The first

pressure chamber wall may be operatively associated with the first converter wall. In order to obtain pressure intensification desired between the thermal energy converter and the pressure intensification means, the surface area of the first converter wall may be significantly larger than the surface area of the first pressure chamber wall facing the pressurised fluid. The first converter wall and the first pressure chamber wall may be connected by a common shaft extending through the first converter chamber and into the first pressure chamber. The first pressure chamber may include the pressure intensification means. Alternatively, the first pressure chamber may be separate from the first converter chamber. The first pressure chamber may include a first pressure chamber wall operatively associated with the first converter wall. The first converter wall is housed in the first [illegible]mber, whereas the first pressure chamber wall may be separate and housed in [illegible] pressure chamber. The first converter wall and the first pressure wall may be linearly or otherwise fixed to one another along their general axes of travel with a first shaft extending through the first converter chamber and into the first pressure chamber. Accordingly, the pressure intensification means may include the first pressure chamber housing the first pressure chamber wall movable to define together a first pressure compartment of variable volume.

The first pressure compartment may be selectively in communication with the accumulator means, whereby to provide the pressurised fluid to the accumulator at elevated pressures. Interposed in the communication lines between the first pressure chamber compartment and the accumulator may be a valve or a combination of valves. The combination of valves may include a first one way outlet valve permitting delivery of pressurised fluid to the accumulator on completion of each stroke or part thereof. The

combination of valves may also include a one-way inlet valve to permit return of non-pressurised fluid from the accumulator via the work output means. The first pressure compartment, the accumulator and the work output means may therefore form a closed system in which the pressurised and non-pressurised fluid is recycled. The one or more valves may be spring loaded ball and socket valves as is standard in the art or may comprise any other suitable valve arrangement effective to perform the required valve functions. The first pressure chamber may further include a first opposed pressure compartment defined by the first pressure chamber and an opposite side of the first pressure chamber wall. The first opposed pressure compartment may be vented to the atmosphere whereby to provide little resistance to the first pressure wall during a stroke cycle. Alternatively, the first opposed pressure compartment may be in communication with a collector vessel whereby any leakage through seals and the like associated with the first pressure wall may be fed back into the system. The pressure intensification means may include return means for urging the first converter wall back to a return position during the second part of the stroke cycle. The return means may include spring bias. Alternatively, the return means may include a piston return chamber including a first pressurised return compartment in communication with a return accumulator to maintain the pressure of the first return compartment at a constant level. The return piston may be operatively associated with the first converter wall and the first pressure wall, for example by means of the first shaft. An opposed return compartment on the other side of the return piston wall may be vented to the atmosphere.

The accumulator may include one or more closed containers for housing the compressed wall pressurised fluid at elevated pressures. The pressurised fluid may be non-compressible. For example, the pressurised fluid may be hydraulic oil. Each of the

closed containers may include a movable accumulator wall. Together with the movable accumulator wall, the containers may define a closed compressible fluid compartment of variable volume. The movable accumulator wall may be flexible. The movable accumulator wall may be elastic. The movable accumulator wall may be in the form of a diaphragm. Accordingly, each container may define a compressible fluid compartment and a non-compressible fluid compartment. The non-compressible fluid compartment may be in communication with the work output means.

The non-compressible fluid may be bled at a regular rate from the non-compressible fluid compartment at elevated pressure to drive the work output means. Alternatively, the work output means coupled to the accumulator may be selectively operable, intermittently operable or programmably operable whereby to bleed o[illegible] at a predetermined or required time and rate. The work output means may generate electricity. The work output means may act as a pump for other fluids, such as water required for domestic or irrigation purposes or sewage effluent. The work output means may include an alternator or a generator. The work output means may charge a battery whereby to store electrical energy.

The thermal energy converter may include a second expansion chamber. The second expansion chamber may be operatively associated with the first expansion chamber. The second expansion chamber may include a second movable converter wall. The second converter wall may be operatively associated with a first converter wall. For example, the second converter wall may include a piston having a second shaft. The second shaft may be operatively connected to the first shaft. The first and second shaft may be connected by means of a rocker arrangement. The first and second shafts may be so arranged as to

reciprocate in opposites directions. The first part of the stroke cycle of the first converter wall may correspond to the second part of the stroke cycle of the second converter wall.

The rocker arrangement may include locking means to provide a dwell time in which the first expansion compartment may accumulate pressure and the first opposed compartment may dissipate pressure to maximise the power of the stroke of the first converter wall. The first expansion compartment may pressurise at the same time as the second opposed converter compartment pressurises whilst the first opposed converter compartment and the second expansion compartment experience dissipation of pressure. Pressurisation may be accomplished by an evaporator and dissipation of pressure may be accomplished by a condenser. A[illegible] pre-determined period, the dwell time may be [illegible] completed and the power [illegible] with the first and second converter chambers executed with a maximum differential pressure existing between the expansion and opposed compartments of the respective first and second converter chambers. The working fluid of the first expansion compartment may be in communication with the working fluid of the second opposed converter compartment via the temperature modifier and the working fluid of the first opposed converter compartment may be similarly in communication with the second expansion converter compartment. The dwell time may be achieved by locking the rocker arrangement in a particular toggle position at the end of each part of a stroke cycle. Alternatively, the dwell time may be achieved by closing the one way valves interposed between the pressure intensification means and the accumulator. Accordingly, where the pressurised fluid is non-compressible, the closing of the one-way valves may be used to effectively lock the first and second converter walls in a particular position. In another arrangement, a pressure switch may be used to gauge when the converter compartment

reaches a predetermined pressure level, whereby to then activate the cylinder or solenoid holding the toggle arrangement in position thereby commencing a new part of the stroke cycle.

The rocker arrangement may be adapted to freely pivot about a rocker axis. There may be single or multiple pistons, shafts and rockers. The rocker arrangement may include one or more active toggle means. The toggle means may include hydraulic or pneumatic ram means or solenoid means. The toggle means may include a ram or solenoid either side of the rocker axis. The rocker arrangement may further include a over-centre arrangement whereby to lock the rocker arrangement in a particular position at the end of each part of each stroke cycle.

The respective stroke cycles of the first and second converter walls may be opposed and complementary. The first part of the stroke cycle of the first converter wall may correspond to the second part of a second stroke cycle of the second converter wall. The second part of the stroke cycle of the first converter wall may correspond to a first part of the second stroke cycle of the second converter wall. The pressure storage means may deliver pressurised fluid through to the accumulator means on completion of each part of each stroke cycle of each of the first and second converter walls. On completion of the first part of the stroke cycle of the first converter wall, the thermal energy converter may be locked against movement for a period of dwell time whilst the working fluid is cooled by the temperature modifier to prepare for a return stroke corresponding to the second part of the stroke cycle.

The first and second converter and opposed compartments may all include working fluid. The dwell time in preparation for the first part of the stroke cycle may involve both

the first converter compartment and the second opposed converter compartment including working fluid heated by the temperature modifier. The dwell time in preparation for the first part of the stroke cycle may also involve both the second converter compartment and the first opposed converter compartment including working fluid cooled by the temperature modifier whereby to maximise the pressure differential between the opposed compartments to optimise the power of the stroke corresponding to the first part of the stroke cycle by the thermal energy converter.

The first and second opposed compartments may be vented and in communication with air at ambient pressure. The first and second opposed compartments may be in communication with a collector vessel in a closed system. The collector vessel may be maintained at roughly atmospheric pressure and may be effective to return pressure back to the first and second converter compartments to reduce the effect of any leakage through seals associated with the first and second converter walls.

The first converter wall may be a piston including a shaft adapted to travel through an aperture in an end wall of the first converter chamber. The aperture may include seal means to reduce undesirable leakage of the working fluid from the first converter compartment through the aperture or the seals associated therewith. The seal means may include a sleeve to encapsulate the shaft adjacent the aperture. The sleeve may be concertinaed and may be in the form of bellows adapted to guard against pressure leakage.

Possible and preferred features of the present invention will now be described with particular reference to the accompanying drawings. However it is to be understood that the features illustrated in and described with reference to the drawings are not to be construed as limiting on the scope of the invention. In the drawings:

Fig. 1 is a schematic side sectional or view of an engine and associated apparatus in accordance with one possible embodiment of the present invention;

Fig. 2 is a schematic side sectional view of an alternative fluid heat exchange and expansion chamber arrangement.

Fig. 3 is a schematic cross-sectional view of a thermal engine converter according to a third embodiment of the invention;

Fig. 4 is a schematic side view of the third embodiment of the invention showing a rocker arrangement with more clarity;

Fig. 5 is a schematic side sectional view of a thermal energy converter according to a fourth embodiment of the invention;

Figs. 5A, B and C are schematic sectional side views of a thermal energy converter according to a fifth embodiment of the invention showing the converter in various stages of a stroke cycle;

Figs. 6A and B are schematic side sectional views of a first converter chamber according to one aspect of the invention;

Fig. 7 is a schematic side sectional view demonstrating the intensification factor in relation to different embodiments of the invention;

Fig. 8 is a schematic sectional side view of a sixth embodiment of the invention.

Referring to Fig. 1, the engine includes an expansion chamber 11 and associated heat exchange means 20 for alternately heating and cooling working fluid 12 in the chamber 11.

The working fluid 12 can be a refrigerant such as a freon, ammonia, isopentanes, AZ20 or any suitable working fluid which experiences substantial pressure variations upon change in temperature. In particular, as the temperature of the refrigerant is increased, the

pressure can increase substantially within the chamber 11. Conversely if the temperature of the refrigerant 12 is decreased, the pressure in the chamber 11 decreases.

The chamber 11 is in the form of a cylinder and the upper wall of the closed chamber in which the refrigerant 12 is located is defined by the piston 13 which is movable vertically within the chamber 11 in response to changes in pressure of the alternately heated and cooled refrigerant 12.

The engine also includes compression means 30 coupled to the piston 13 for compressing a storage fluid 32 located within the compression cylinder 31 of the compression means 30. The piston 33 which is movable in the compression cylinder 31 is coupled by shaft 35 to the piston 1[illegible] expansion chamber 11 so as to be moved by movement of the piston 13.

The storage fluid 32 in the cylinder 31 when being compressed by piston 33 flows to the accumulator means 40 for holding the pressurised storage fluid at an elevated pressure and at progressively increasing pressure as the compression means 30 cycles in response to cyclical movement of the piston 13 of the expansion chamber 11.

Associated with the accumulator means 40 is a controlled work output system 50 for utilising the stored pressurised storage fluid to perform useful work.

Basically, the pressure increase that occurs with increasing temperature of the working fluid 12 is utilised to force the piston 13 to move within the cylinder 14 defining the expansion chamber 11 which, in turn, moves the fluid compression piston 33 to compress storage fluid 32 in the cylinder 31, this pressurised storage fluid 32 being supplied to the accumulator means 40 which functions to hold the stored energy as hydraulic fluid pressure. The stored hydraulic pressure can be released in a controlled manner through the controlled

work output system 50 which, in the illustrated embodiment, includes an hydraulic motor 51 coupled to an alternator or generator 52 to generate electrical power, which can be used for example to charge electrical batteries 53 or by controlled release of stored pressurised hydraulic fluid from accumulators, will maintain constant revolutions of the hydraulic motors coupled to the generator or alternator thereby producing a fixed frequency and voltage output to be fed directly to an electrical grid or to a customer. Alternatively frequency inverts could be used to maintain constant frequency or voltage. It is also possible to have multiple cylinders with pressure release valves to regulate hydraulic flow, and to run hydraulic motors directly without accumulators.

To [illegible]er explain details of the engine and its operation, the piston 13 in the [illegible] ex[illegible] 11 as illustrated is at or near the end of its return stroke at the end of a cooling cycle or about to begin a heating and compression cycle. The engine in Fig. 1 has associated therewith an hydraulic fluid accumulator 45 which is part of a piston return mechanism 44 operative to return the piston 33 and piston 13 at the end of a heating and compression cycle when the storage fluid 32 has been compressed and supplied to the accumulator means 40. The accumulator 45 is coupled to a piston return cylinder 46 in which there is a piston 47 coupled to piston 33 of the compression means 30. At the start of and during the return cycle of the pistons 13 and 33, the elevated pressure of hydraulic fluid 48 within the upper portion of the cylinder 46 supplied from the pressure accumulator 45 acts on the piston 47 to return the piston 33 and piston 13. Conversely during the compression stroke, piston 47 is moved upwardly in the cylinder 46 to elevate the pressure of the hydraulic fluid 48 which is returned through valve 49 to the pressure accumulator 45. As an alternative to this piston return means 44, it may be possible to provide a compression

spring within the cylinder 31 acting on the upper face of the piston 33, the spring being operative to apply downward force to return the piston 33 and piston 13 during the return cooling stroke or cycle of the engine.

The heat exchanger means 20 provided in the expansion chamber 11 which is operative to alternately heat and cool the working fluid 12 has an associated means 22 for cycling the fluid heating and fluid cooling functions. The cycling means 22 is illustrated as a valve 23 (and associated timing or switching mechanism - not shown) which is operative at or about the point in the cycle illustrated in Fig. 1 to open to allow heating fluid to enter the heat exchange coil 21 from the heating fluid inlet 24. The flow of heating fluid from the inlet 21 through the cycling means 22, illustrated as a two-way valve 23, and through the heat exchange coil 21 raises the temperature of the working fluid 12 within the expansion chamber 11. If the working fluid 12 for example comprises a refrigerant such as refrigerant type R409a, and the temperature of the refrigerant was raised to 42°C, the pressure of the heated refrigerant would be 935.9 kPa (137psi). If the diameter of the piston 13 is, say, 304mm (12 inches) the surface area of the piston would be 113 square inches providing a force of 15494 lbs acting on the lower face of the piston 13.

With piston 13 coupled as shown in Fig. 1 to the piston 33 of, say, about 4 inches diameter, the piston 33 can create an hydraulic force of perhaps 2000psi to act on the storage fluid 32 in the cylinder 31.

There are obviously many variables in the arrangement of the engine that affect performance parameters including the respective piston areas, the strokes of the respective pistons, whether one directly couples pistons as shown using shaft 35 or whether indirect coupling, e.g. through leverage to magnify pressures, is used, the number of cycles of

heating and cooling achievable, the temperatures to which the working fluid is heated and cooled, and the nature of the refrigerant working fluid. It is also possible to have multiple expansion chambers 11, multiple associated heat exchange means 20, and multiple compression means 30 operating out of phase with each other in parallel so that substantially continuous flow of pressurised hydraulic fluid occurs instead of the intermittent flow to the accumulator means 40 that occurs in the single cycle engine illustrated. The pressurised storage fluid 32 being output in line 36 is preferably passed directly to the hydraulic accumulators 41, 42 so that these accumulators store the hydraulic fluid at high pressure. The hydraulic fluid can be released in a controlled manner at a constant flow rate to the output system 50.

When the pistons 13 and 33 have reached the tops of their stroke[illegible]ge [illegible] fluid 32 has been expelled from the cylinder 31 through line 36 to th[illegible]eans 40, various valves can be switched as required to commence a cooling return stroke. In particular, the valve 37 in output line 36 can be closed, and the valve 57 through which low pressure hydraulic fluid can be returned to the cylinder 31 is opened to allow hydraulic fluid from storage reservoir 55 to flow through line 56 to refill the cylinder 31 as the piston 33 moves downwardly. To promote or assist return of the pistons 33 and 13 in their respective cylinders 31 and 14, pressurised hydraulic fluid in accumulator 45 can flow through valve 49 to the piston return cylinder 46 to urge the piston 47 downwardly.

At substantially the same time, the two way valve 23 is switched to pass cooling fluid from inlet 26 through the valve 23 and into the heat exchange coil 21 so as to cool the refrigerant 12 within the expansion chamber 11. With the particular exemplified refrigerant

are K409a, if the temperature of the refrigerant 12 is reduced to 18°C, the pressure drops to 418.9 kPa (61.57psi).

When pistons 13, 33 and 47 have reached the bottoms of their strokes, valve 57 is closed because cylinder 31 is now charged with new hydraulic fluid, valve 37 is opened, and two-way valve 23 is switched again to admit heating fluid to the heat exchange coil 21.

The heating and cooling fluids supplied respectively through inlets 24 and 26 may be obtained from any convenient sources. Because the temperature to which the refrigerant 12 is heated can be relatively low (42°C in the example), the possible sources of heating fluid can include solar energy fluid heaters. For example, water passing through polypropylene or other suitable piping exposed to solar radiation can be used as the source of heating fluid. Other possible sources of heating fluid ca[illegible]ermal sources such as bore water [illegible] being drawn from underground aquifers [illegible]aperature. Another example of a source of heating fluid can be heated exhaust gases from an engine or from industrial plant and equipment. The exhaust gases may be passed through the heat exchange coil 21 directly. Alternatively, water heating coils can be passed in heat exchange relationship to exhaust pipes, chimneys, or the like carrying the heated gases so that the water is heated thereby and the heated water is used as the heating fluid passing through the coil 21 in the engine.

Likewise, the cooling fluid passed through the coil 21 during the cooling return cycle can be any suitable or conveniently available cooling fluid, such as cool or cold water from any convenient available source.

In the modified embodiment illustrated in Fig. 2, the heat exchange means 120 is substantially a separate arrangement from the expansion chamber 11 although the working

fluid 12 is in direct communication with the expansion chamber 11. In this embodiment, the heat exchange means 120 includes a thermal fluid chamber having an inlet 124 through which heating fluid and cooling fluid are alternately introduced under control of a cycling means such as the cycling means 22 in the embodiment of Fig. 1. The heating fluid and cooling fluid pass through the thermal fluid chamber 121 and exit through outlet 125. The chamber 121 is substantially totally enclosed within an insulating jacket 126 to reduce thermal losses through the outside walls of the chamber 121. Also the outside walls of the chamber 121 may be relatively thin material to reduce thermal load losses in alternately heating and cooling the walls. As well as thin metallic walls, the walls could be non-metallic, e.g. ceramic or alumina material to have low heat conductivity.

Located [illegible] thin the chamber 121 is a side chamber 130 of the expansion chamber 11, [illegible] ber 130 being in communication with the expansion chamber 11 so as to be filled with the working fluid 12. The side chamber 130 has highly heat conductive walls 131 so that heat from heating fluid passing through chamber 121 from inlet 124 to 125 is rapidly supplied to the working fluid 12 within the chamber 130. This in turn will cause the desired rise in pressure in the expansion chamber 11 to move the piston 13 during the heating and storage fluid compression stroke. Conversely, when cooling fluid is being passed through inlet 124, through the chamber 121, and out of outlet 125, the heated working fluid 12 in the side chamber 130 rapidly yields up heat through the walls 131 to the cooling fluid to reduce the pressure in the expansion chamber 11 acting on the underside of the piston 13 during the cooling return stroke.

It will be seen that the engine is relatively simple in operation and many of the components can be readily available existing types of equipment. The construction,

assembly, installation, operation and maintenance of the engine and associated equipment can be relatively simple, and operators and maintenance personnel need not be highly educated and trained personnel since the technology is relatively simple.

Turning to Fig. 3, there is shown a thermal energy converter 140, a temperature modifier 150, a pressure intensification means 160 and a rocker arrangement 170, being components of an engine 135 for converting thermal energy. The accumulator 40 and controlled output means 50 of the engine 135 are as described in relation to the embodiment of Fig. 1.

The thermal energy converter 140 includes a first expansion chamber 141 and a second converter chamber 142. The first converter chamber 141 houses a piston 143 adapted for linear reciprocal travel within the first converter chamber 141. A first shaft 144, extends axially either side of the first piston 143, is fixed relative to the piston 143 and extends through apertures in the opposed ends of the cylindrically shaped first converter chamber 141. The piston 143 divides the first converter chamber 141 into two compartments, the first expansion compartment 145 which varies in volume in inverse relation to the volume of a first opposed converter compartment 146 in the first converter chamber 141.

Co-axially fixed to the first shaft 144 is a first pressure piston 161 housed in a first pressure chamber 162. A similar mirror image arrangement may be seen in relation to second pressure chamber 166. The first pressure chamber 162 defines, together with the first pressure piston 161, first pressure compartment 163 and first opposed pressure compartment 164. First opposed pressure compartment 164 is vented to the atmosphere.

First opposed pressure compartment 163 is in communication with the accumulator means 40 described in relation to Fig. 1 via a pair of inlet/outlet valves 165.

The temperature modifier 150 includes a condenser 151 and an evaporator 152 in communication with the first and second converter chambers 141, 142 via separate lines: condenser line 153 and evaporator line 154. The inline condenser 151 includes a coil which extends through cool fluid, such as water from a local stream or another cool water source which flows to cool the working fluid in condenser line 153. The evaporator 152 also includes a coil through which hot fluid, such as hot water or gas from a local source, flows to heat the working fluid in evaporator line 154. The working fluid is refrigerant AZ20. Inline evaporator and condenser valves 155, 156 are located inline in the condenser and evaporator lines 153, 154.

The second converter chamber 142 and the second pressure chambe [illegible] reflectively the mirror image of first converter chamber 140 and first pressure chamber 160, respectively. Accordingly, extending through the second converter chamber 142 and into the second pressure chamber 166 is a second shaft 147 attached at a second end to a rocker 171. Similarly, the lower end of the first shaft 144 is pivotably attached to a first end of the rocker 171. In the lower portion of Fig. 3, the rocker arrangement 170 is shown and is shown in greater detail in Fig. 4. The rocker arrangement 170 includes rocker 171 adapted to pivot about pivot point 172. The pivoting of rocker 172 is controlled by a co-axially pivotal toggle bracket 173 whose pivoting action is controlled by an over-centre arrangement 174 which, together with a pair of hydraulic cylinders 176 pivotally extending from a base 175 to an end each of the toggle 173, enables the rocker arrangement to be locked into position until the pair of cylinders 176 are again activated. Screw bolts 177 are

threaded into the base 175 to provide adjustable stops limiting the extent of travel of the over-centre 174. By this mechanism, a dwell time is achieved for the engine of Fig. 3 or Fig. 4 whereby to permit pressure build up of the working fluid in complementary compartments of the first and second converter chambers 141, 142. This optimises the pressure differential between the first and second expansion compartments and the first and second opposed converter compartments to maximise the power strokes associated with each of the first and second converter chambers. Note that the power stroke in the arrangement shown in Fig. 3 is effected in both the first and second converter chambers 141, 142 simultaneously, whereas in the arrangement shown in Fig. 4, the first and second opposed compartments 136 are vented to the atmosphere or to a pressure collector, whereby each converter chamber provides a power stroke [illegible] part of the stroke cycle.

Whereas the arrangement of Fig. 3 is co[illegible]ce greater power compared to that of Fig. 4, it is also less efficient in its use of the available energy as the resistance provided by the opposing compartment during a power stroke must overcome the ambient pressure of the refrigerant to effect the power stroke. However, both chambers may perform the power stroke at the same time.

With particular reference to Fig. 3, in a first part of the stroke cycle, first expansion compartment 145 is cooled to lower the pressure to the ambient pressure of the refrigerant, that is about 200psi, whilst dwell time achieved by locking rocker arrangement 170 permits the pressurisation of the first expansion compartment 145 as well as the corresponding second opposed converter compartment 148. This is achieved by opening the evaporator valves 156 to the second opposed converter compartment 148 and the first expansion chamber 145 and closing the condenser valves 155 so that the working fluid achieves

pressures of between 500 and 800psi in these compartments 145, 148, whereas the first opposed converter compartment 146 and a second expansion compartment 149 are cooled by exposure of the working fluid in line 153 to the cooling fluid of the condenser 151. Evaporator line 154 and condenser line 153 both include small piston pumps adapted to circulate the working fluid through lines 153, 154. The piston pumps (not shown) require minimal energy and may therefore involve a separate feed line from the accumulator or be driven by a battery charged by the work output means or directly from generator or alternator output.

In Fig. 4 it is clearly shown that the first and second converter chambers are pivotable about first and second pivot points 178, 179 to ensure that the first and second shafts 144, 147 follow the cur[illegible] by their pivotal attachment to the rocker 171.

In Figs. 5A, [illegible]s shown an alternative arrangement in which the first and second opposed converter compartments 180, 181 are vented to the atmosphere or are in communication with a pressure collector vessel which is adapted to feed escaped pressure back into the working fluid system 182. In this regard, leakage is likely to occur through the annular seals 183 about the pistons 184, 185. Shown in Figs. 6A and B are bellows 190 which are in the form of concertinaed sleeves surrounding sections of the shaft 191 either side of the converter chamber 192. The bellows 190 may be in communication with pressure collector vessel (not shown) to reduce pressure loss from the overall system.

Referring to Figs. 7A and B, the schematic diagrams illustrate how pressure intensification is achieved in the transition from the expansion compartment corresponding to a relatively large Area "Y" to the smaller piston head of the pressure intensification means (Area "X"). In Fig. 7B, a central cylindrical column 193 is located on the single piston head

194 and the pressure chamber 195 corresponds to the opposed converter compartment, such that the thermal energy converter and the pressure intensifier are housed in a single chamber. In each example shown, a pressure intensification factor of 3 is achieved by different thermal engine converter/pressure intensifier combinations.

In Fig. 8 there is shown a generator arrangement comprising a thermal energy converter 200 interposed between a pair of pressure chambers 201, 202 whereby each part of the power stroke cycle of converter 200 alternately charges a different pressure chamber 201, 202. In such an arrangement, provision of a dwell time is desirable to achieve optimum pressure differential between converter compartments 203, 204.

It will be seen also that the engine can use heating and cooling sources available at many places around the world including over great ranges of climates since it is the temperature differential between the temperature to which the working fluid is heated and to which it is cooled that determines the possible work output rather than the absolute values of those temperatures. Also, the engine can be particularly environmentally acceptable since combustion of fuel is not necessary for its operation and/or sources of heat that is currently being wasted or lost (e.g. in heated exhaust from engines and industrial plant and equipment) can be utilised to perform useful work.

It is to be understood that various alterations, modifications and/or additions may be made to the features of the possible and preferred embodiment(s) of the invention as herein described without departing from the spirit and scope of the invention.

Dated this 26th day of March 2002

PATENT ATTORNEY SERVICES

Attorneys for

RICHARD LAURANCE LEWELLIN

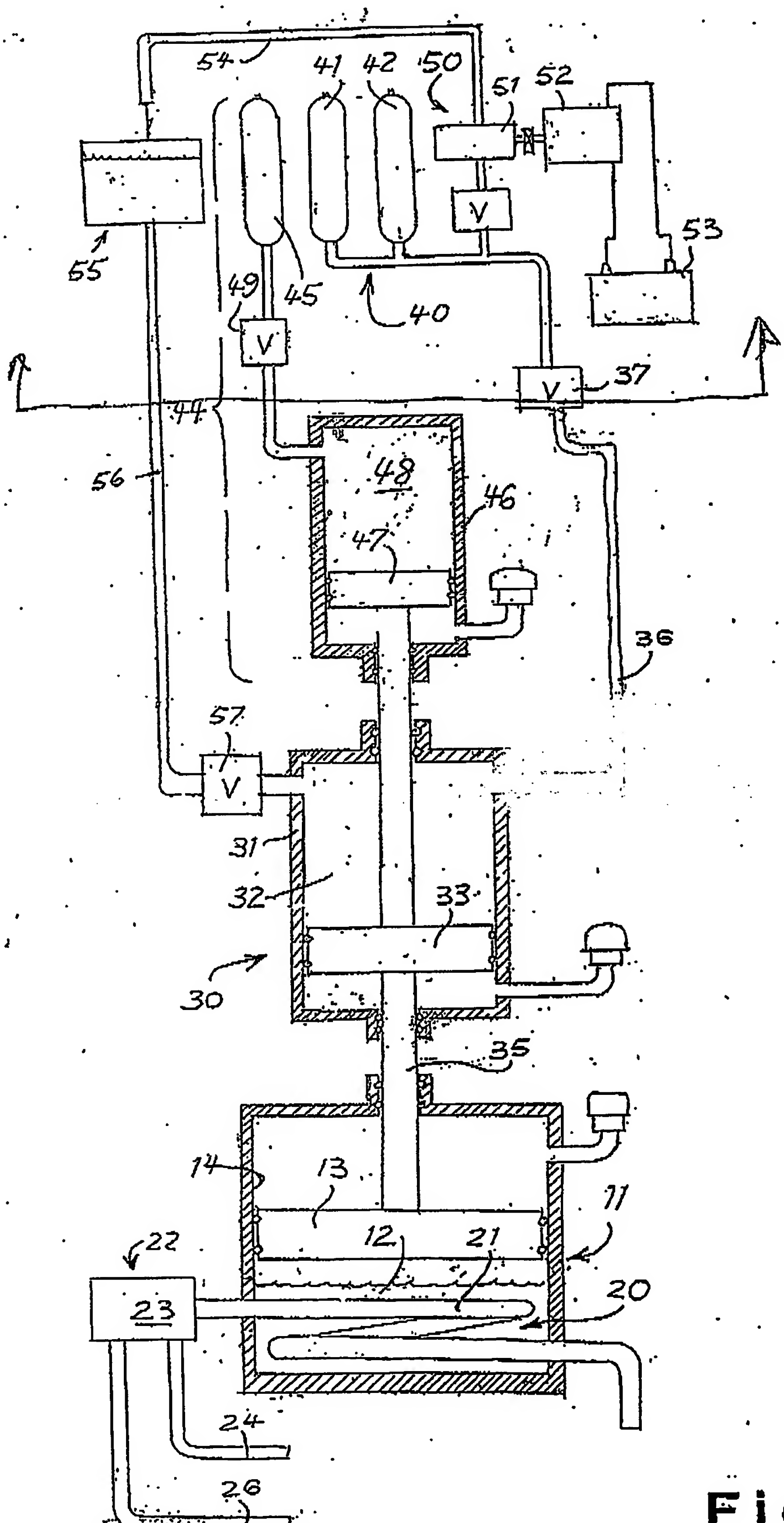
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Fig. 1

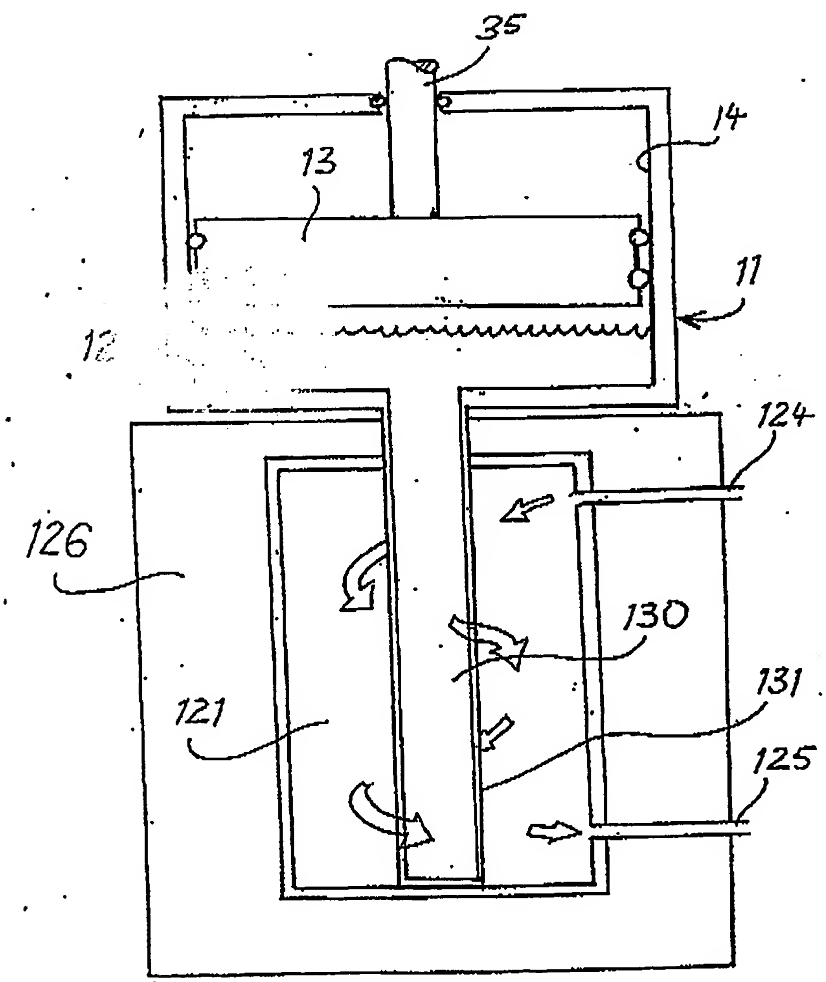
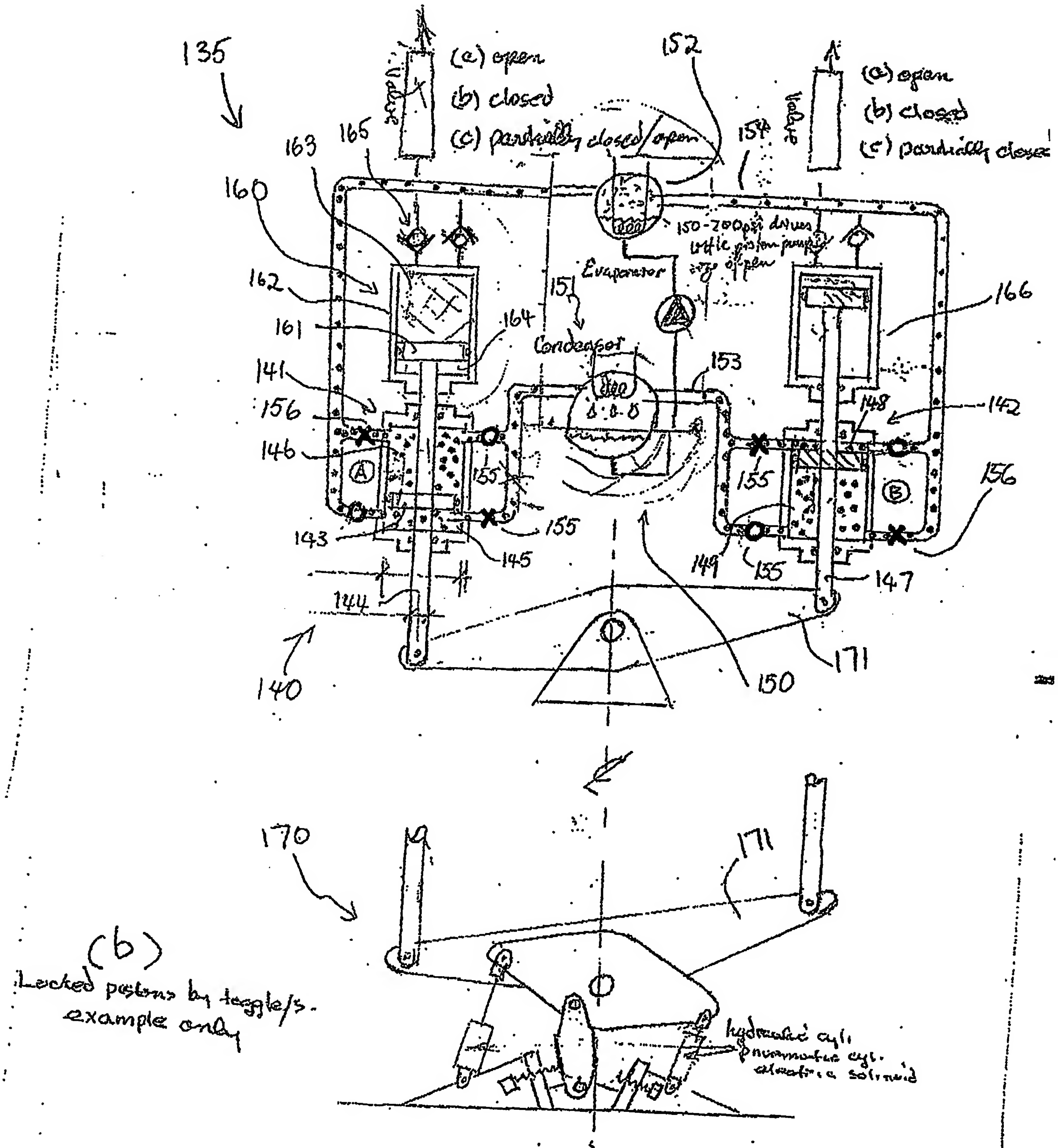
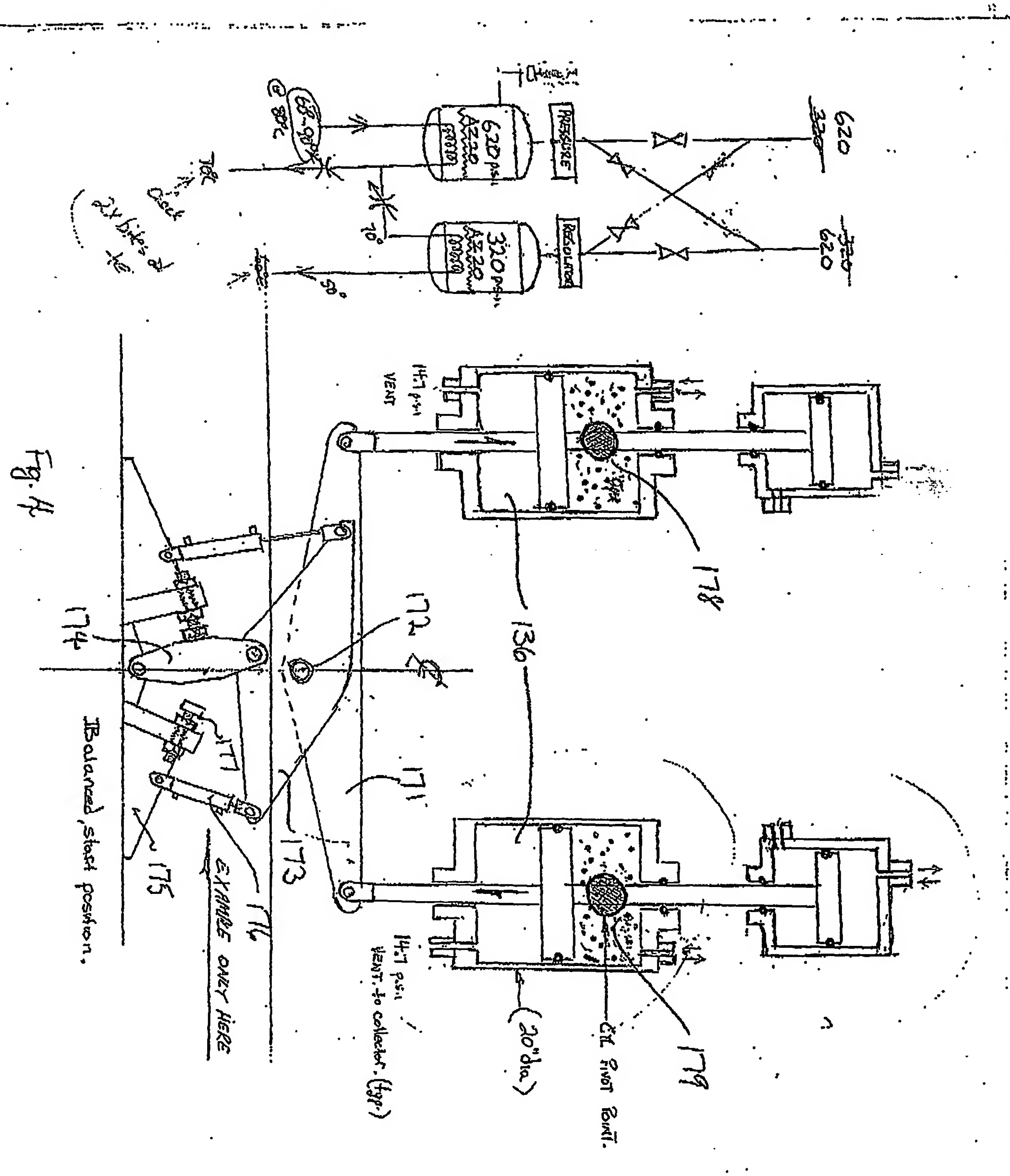


Fig.2



DWELL CONTROL

Fig: 3


620
320
620
320
PRESSURE
REGULATOR
620 p.s.i.
320 p.s.i.
14.7 p.s.i.
VENT
178
136
172
174
171
177
173
175
176
179
C/L PIVOT POINT.
(20" dia.)
14.7 p.s.i.
VENT. to collector. (typ.)
EXAMPLE ONLY HERE
Balanced, start position.
Fig. A

Leaverage intensification System

<u>AZ20 Refridgerant</u>

80°C ⇌ 5800 kPa. 852.6 p.s.i

piston "A" = 12" dia _ area = 113.09 sq inch
7 dia = 2" dia - area = 3.1416 sq inch
= effective piston area = 109.95 sq inch.

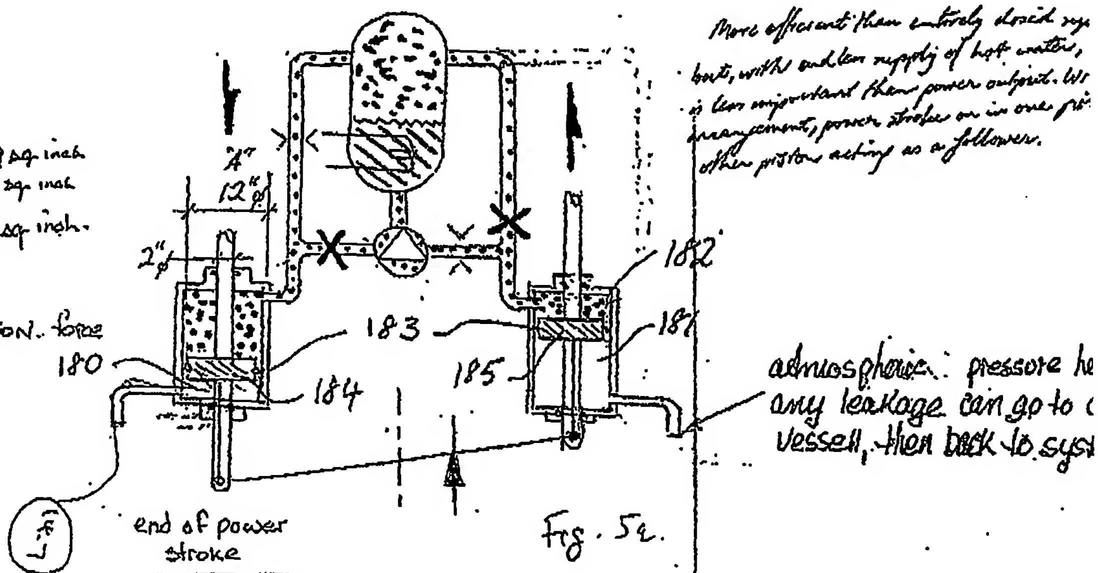
109.95 sq inch × 852.6 p.s.i

$= \frac{93743 \text{ lb}}{2240} = 41.84$ TON. force

Leaverage intensification =

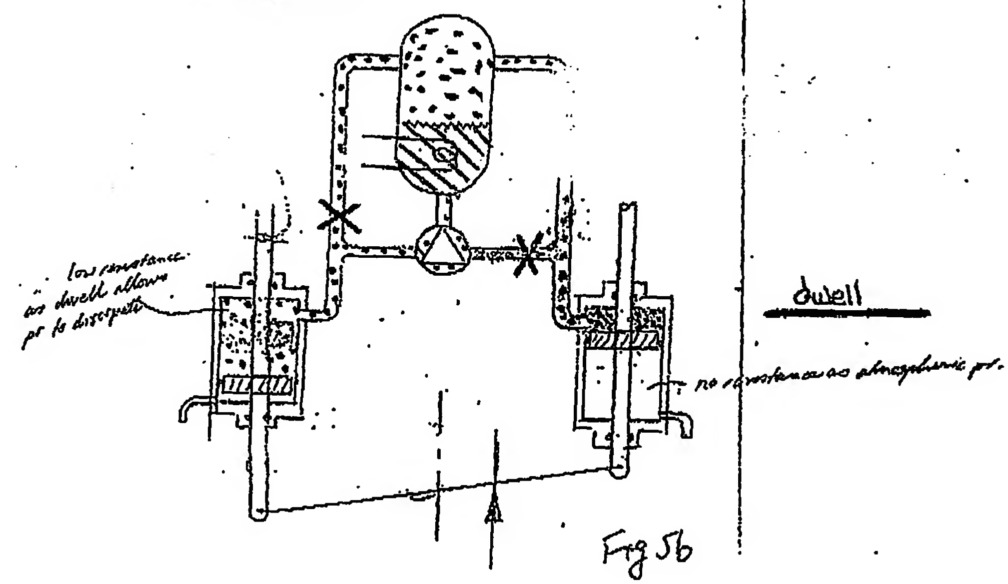
2PE and AZ20 had
pressure of 200 p.s.i

More efficient than entirely closed sys
but, with and less supply of hot water,
is less important than power output. W
arrangement, power stroke on in one pi
other piston acting as a follower.



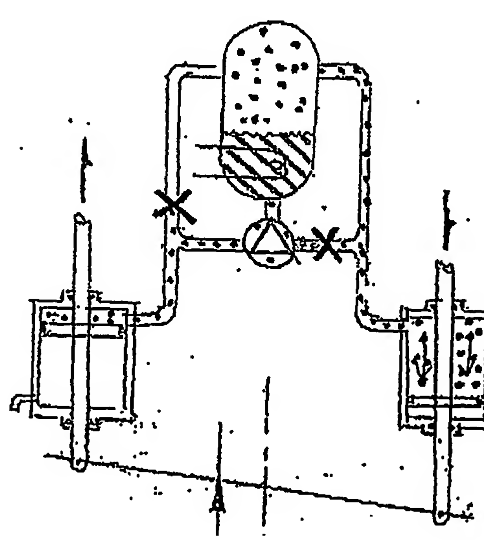
end of power stroke

Fig. 5a.



dwell

Fig 5b



Power stroke

Fig. 5c

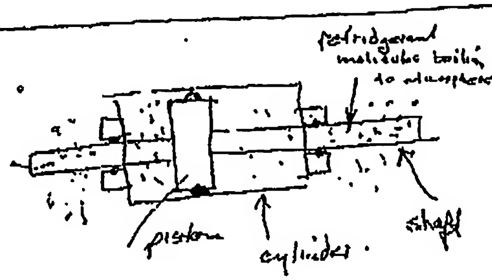


Fig. 6a

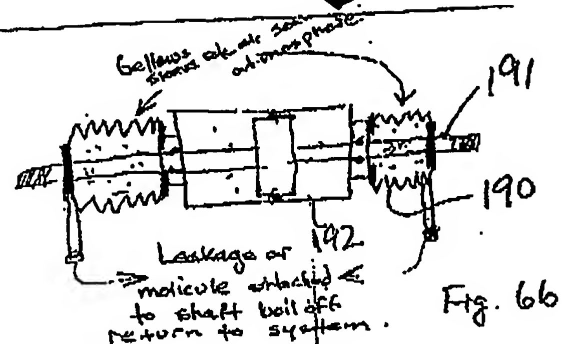


Fig. 6b

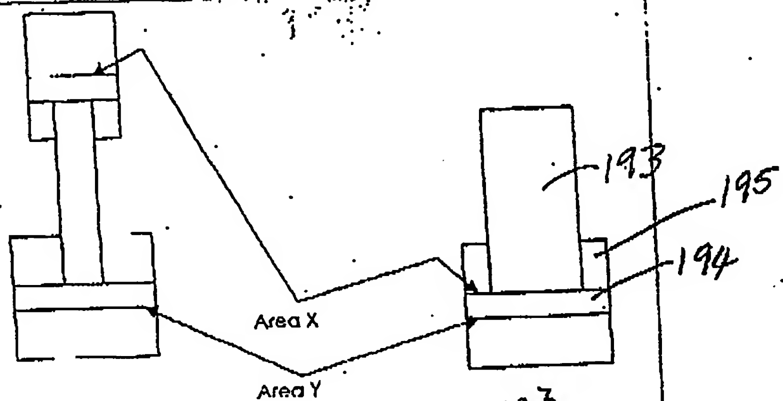


Fig 7A

fig 7B

Inten... factor = $\frac{\text{Area Y}}{\text{Area X}}$

Eg ... Area X = 4

Intensification factor

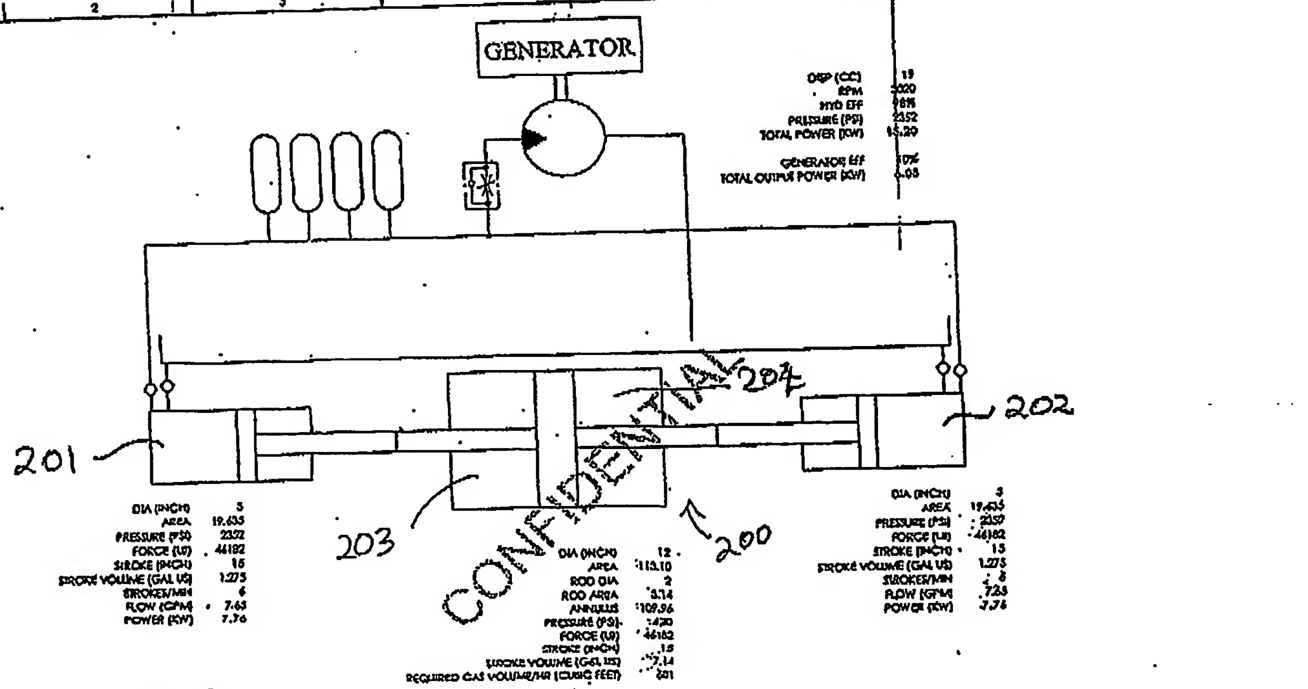


Fig 8

REV | DATE | DESCRIPTION

SOUTHERN HYDRAULICS PTY LTD
HYDRAULIC EQUIPMENT DESIGN & MANUFACTURE
PORT MELBOURNE, VICTORIA AUSTRALIA

DESIGNER:
SCALE:
SIZE: